**<u>Report No: MIT-GFR-014</u>**

## Topical Report

# 300 MWe Supercritical $CO_2$ Plant Layout and Design

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## 1. Introduction

### 1.1 General

This topical report summarizes Year 1, Task 2 progress under the contract "Qualification of the Supercritical $CO_2$ Power Conversion Cycle for Advanced Reactor Applications", having the scope defined in the following excerpts from the Sandia statement of work:

**Objective**

To complete assessment of the Supercritical $CO_2$ Brayton Cycle to the point where confident commitment can be made to its accelerated development for use in advanced reactor concepts.

**Task 2.**

Engineer a plant layout for a 300 MWe power train and develop a first order cost estimate for this power plant, focusing on savings due to its simplicity and compactness.

**University will have the following goals:**

**Task 2.**

Aim at quantifying the postulated cost advantage of the S-$CO_2$ cycle, which so far has been asserted mainly on the basis of engineering judgment. This is an essential task since there is a broad consensus that capital cost reductions on the order of thirty percent or so are necessary for nuclear power to make major inroads into the future market for central station generation of electricity.

### 1.2 Scope of This Report

This topical report focuses exclusively on plant layout and cost assessment for the supercritical $CO_2$ Brayton power cycle. As such it is somewhat specialized and presupposes familiarity with considerable more general background material. Such is available in the recent comprehensive topical report:

V. Dostal, M.J. Driscoll, P. Hejzlar, "A Supercritical Carbon Dioxide Cycle for Next Generation Reactors," MIT-ANP-TR-100, March 10, 2004

Note that this report covers much the same material as recently reported under Task 2 of the project's annual report:

MIT-GFR-012, "Annual Report: Qualification of the Supercritical $CO_2$ Power Conversion Cycle for Advanced Reactor Applications", by Y. Wang et al, April 9, 2004

## 1.3 Background

Before addressing the specifics of the current reference design, review of some background is appropriate. In particular, the reasons for specifying 300 MWe as the standard power train rating need to be appreciated. Considerations leading to this specification were as follows:

1. Realization of economy of scale to the extent practical
2. Matching the potential future reactor market, which might encompass individual power plants ranging from 300 to 1200 MWe
3. Taking advantage of modularity, factory fabricability, and transportability
4. Synergism with relevant industrial experience, both nuclear and non-nuclear, and with concurrent GEN-IV reactor development programs

A brief synopsis of relevant factors in each category follows.

Schlenker (1.1) gives scaling relations for (helium) turboset costs. For example, at 12 MPa one has:

$$\text{Cost} \approx (\text{Power})^{0.68}$$

which suggests as large as possible a rating as permitted by other constraints.

Choosing 300 MWe as a rating allows us to follow the PWR precedent of using 1 to 4 loops to compete in both small and large markets. At the small end, for example, the IRIS PWR concept is rated at about 300 MWe and is being designed with the small-grid user in mind. Similarly the GT-MHR is rated at 285 MWe. At the large end one has near term next-generation competitors such as AP-1000, rated at 1000 MWe.

Practical upper-limit size constraints of note are pressure vessel fabrication, where PWR vessels of about 5 m OD at 15 MPa are currently being produced; and universal transportability, where Schnabel rail cars can move loads of several hundred tons, and on the order of five-meter diameters. Respecting these constraints would allow fabrication, repair, refurbishment, and uprating of power conversion units in a factory setting, with attendant savings. We are also favored in this regard by the inherently compact nature of the S-$CO_2$ turbomachinery and heat exchangers of the PCHE (Heatric™) type. Dostal estimates that an integral, all-in-one, S-$CO_2$ power conversion unit (PCU) would be only 54% the volume of a GT-MHR He PCU unit of the same rating.

Synergism with industrial experience is also relevant to our sizing decision. The largest fossil-fired industrial heavy-duty gas turbines are in a comparable size range: (1.2) (e.g. the ABB GT26 unit at 254 MW, the GE MS900G at 282 MW, and the Siemens-Ansaldo V94.3A at 240 MW), when used alone or in combined cycle applications. This provides a ready-made source of balance-of-electrical-plant components for generation and power conditioning. Their materials and bearing technology experience is also relevant. Another useful reservoir of transferable technology in the size range of current interest are the existing supercritical steam, high pressure stage, turbines used in fossil-fired-plants at up to 30 MPa and 600°C (1.3). Materials, bearings and shaft seals are of particular relevance.

Apart from issues associated primarily with unit power rating, we were strongly influenced when it came to component layout by current helium Brayton cycle work—the GT-MHR in particular. Thus in what follows, an integral power conversion unit which bundles all turbomachinery and heat exchangers into a common pressure vessel is the starting point for our further efforts. On an even more basic level, the so-called recompression version of the generic category of S-$CO_2$ cycles – of which some half dozen exist –has been downselected as our reference concept. See Dostal et al (1.4) for a comprehensive review of the considerations leading to these decisions.

## References for Chapter 1


1.1 H.V. Schlenker, "Cost Functions for HTR Direct Cycle Components", Atomkernenergie (ATKE), Bd. 22, Lfg. 4, P. 226, 1974

1.2 N.V. Khartchenko (Ed), "Advanced Energy Systems", Taylor and Francis, 1998

1.3 D. Bittermann, J. Starflinger, T. Schulenberg, "Turbine Technologies for High Performance Light Water Reactors", Proceedings of ICAPP '04, Pittsburg, June 2004

1.4 V. Dostal, M.J. Driscoll, P. Hejzlar, "A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors", MIT-ANP-TR-100, March 2004

# 2 Reference Design Plant Layout and Cost Assessment

## 2.1 Introduction

A principal motivation for giving serious attention to adoption of the supercritical $CO_2$ Brayton power cycle is its prospect for cost reduction in generic GEN-IV applications. As documented in Ref. (2.1), reasons for this expectation include competitive thermodynamic efficiency (e.g. 45% at a turbine inlet temperature of 550°C), simplicity (no need for intercooling or reheating, amenability to single shaft configuration) and extremely compact turbomachinery. To quantify the margin of advantage a necessary first step is to better define an actual plant layout.

## 2.2 Plant Layout Selection

Now that all basic features of the supercritical $CO_2$ Brayton Cycle have been finalized, it is appropriate to become more specific on physical layout of the components. To aid in this endeavor we have surveyed several of the newly proposed thermal and fast gas-cooled reactor Brayton cycle plant designs advanced by various international proponents. Table 2.1 summarizes features pertinent to present interests.

The GCFRs designed in the 1960-1980 time frame were generally coupled to a Rankine cycle. Brayton cycle versions were less well studied. Furthermore, all such GCFRs employed PCRVs, and their helium power conversion units were accommodated inside the PCRV in a direct cycle arrangement. Hence their precedent is of limited value for our present interests.

Table 2.1 also shows the current status of MIT S-$CO_2$ project conceptual designs for power conversion cycle layout. As evident, we have evolved from Dostal's monolithic vertical layout to consider a horizontal integral arrangement; even so, further work is still required to assess variations on and alternatives to this configuration. Considerations leading to these choices are as follows:

**Table 2.1 Representative Recent Nuclear Powered Gas Turbine Plant Layouts**

| Concept | Arrangement / Layout |
|---|---|
| GTHTR 300 (Japan) | • Turbine / compressor / generator encapsulated in horizontal pressure vessel<br>• Recuperator / precooler encapsulated in separate vertical pressure vessel |
| ESKOM PBMR (South Africa) | Three vertical vessels, connected by co-axial ducts; generator outside turbine vessel |
| GTMHR (US/GA, Russia) | Vertical Pressure Vessel enclosing Turbine / HP & LP compressors in central cylinder, precooler/intercooler/recuperator in surrounding annulus; generator in vessel extension (see Fig. 2.1) |
| MIT PBMR | Fully dispersed among a total of 21 railcar/truck-shippable modules: e.g. six recuperator modules. |
| MIT/INEEL LDRD (Dostal) | Single vertical PCU vessel housing all S-$CO_2$ components, with generator outside vessel |
| MIT, This Report | Single horizontal PCU vessel housing all S-$CO_2$ components; separate generator |

**Horizontal Layout**

A prime consideration for this election is the issue of maintainability. With a horizontal PCU vessel one can access both ends. Combined with the use of a segmented vessel and air-slide/rail-guided head removal gear this can largely offset the disadvantage of inaccessibility inherent in packaging all components into a compact integral arrangement.

Another motivation is that horizontal bearings can be employed throughout, to take advantage of the large experience base accrued on combined cycle gas turbine and steam turbine power conversion units.

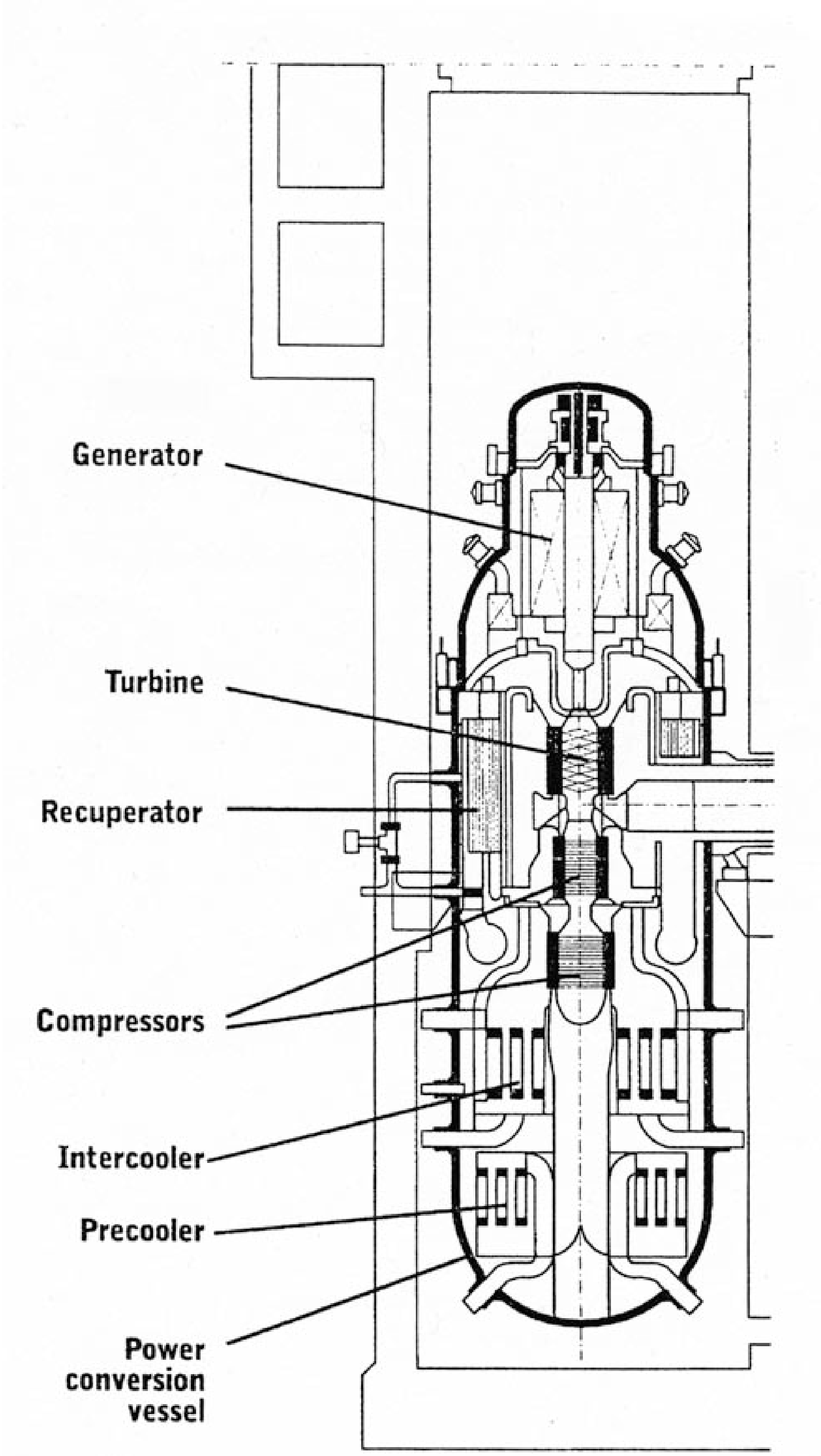


**Fig. 2.1. Main Components of GT-MHR**
**(the arrangement most similar to Dostal's)**

**Separate Generator**

Accessibility and maintainability are also reasons for this choice.

Equally important is the consideration that a separate generator can employ conventional methods for rotor and stator cooling (hydrogen and water, respectively) without undue concern over ingress of these fluids into the power cycle $CO_2$, or vice versa.

Another point is that shaft sealing should be easier for $CO_2$ than for He, and a less stringent leakage specification can be tolerated because $CO_2$ is so inexpensive. Substitution of a shaft coupling/seal unit for a high-pressure encapsulation vessel will also reduce capital cost.

**Integral Configuration**

This is the single most important branch point in the decision tree leading to selection of design features. As evident in Table 2.1, other design teams have populated the full spectrum of choices, ranging from everything in a PCU vessel (GT-MHR) to a multi-module, fully-dispersed arrangement (MIT-MPBR). Supporting considerations for a single horizontal vessel housing the turbomachinery and heat exchangers are as follows:

**1. Physical Feasibility**

Supercritical $CO_2$ turbomachinery is extremely compact, of small diameter and length: e.g. the turbine and both compressors are only a meter or so in diameter. Furthermore the cycle is a simple one, without intercooling and amenable to use of a single shaft configuration.

Even more important is the adoption of Heatric™ PCHE for the recuperators and precooler. Their extremely compact nature, short channel length in particular, enables their emplacement in an annulus surrounding the turbomachines, in an outer vessel comparable in size to a PWR pressure vessel.

**2. The Elimination of Complicated Ductwork**

It is quite difficult to configure high pressure (20 MPa) ducts linking even the small number of components in the S-$CO_2$ cycle. Accommodation of differential and transient temperature gradients dictates use of duct lengths and bend radii larger than mere physical separation would require. This also increases parasitic pressure drop more than one would prefer, even given the fairly tolerant nature of the S-$CO_2$ cycle in this regard. Furthermore, to meet ASME code requirements, the temperature of the high-pressure ducting must be reduced, requiring internal insulation and external cooling—all of which adds to complexity, cost and increased auxiliary power consumption.

## 3. Synergism with Convenient Containment Concepts

Our current reference design for GFR indirect cycle applications encapsulates the entire primary system inside a Prestressed Cast Iron Reactor Vessel (PCIV), which is surrounded by a proximate containment building (vertical cylinder) of modest volume to insure significant equilibrium pressure in the remote likelihood of a LOCA event. This will make decay heat removal feasible solely by natural convection (again only needed if active shutdown cooling fails). To house the power cycle a separate horizontal vessel is used: an arrangement which harkens back to the Shippingport PWR. For our original 600 MWth (250 MWe) design a single loop is employed; for a 2400 MWth (1000 MWe) unit four 600 MWth loops are employed: one each in four cylindrical containments. Figure 2.2 is a photograph of a model showing such an arrangement.

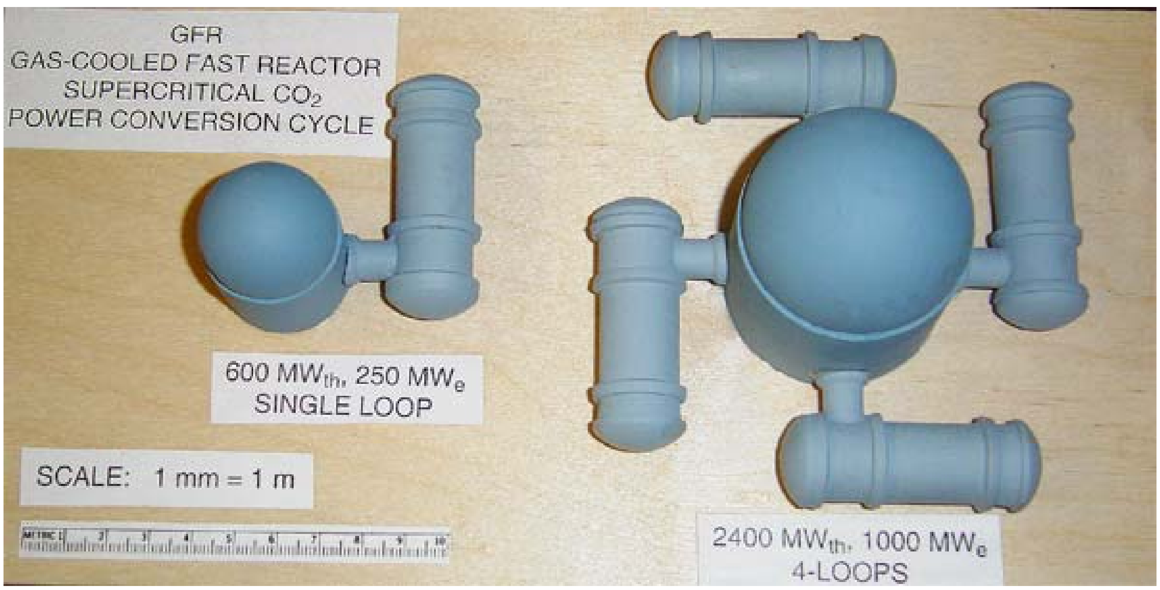


**Fig. 2.2 Model of Single and Four-Loop Reactor Arrangements**

Note that for an indirect cycle, the power conversion unit “containment” vessels can be designed for service at one atmosphere, whereas in a direct cycle they would have to accommodate approximately five atmospheres.

A final, but practical, reason for using an integral configuration is that cost estimates for the GT-MHR, to the extent that such are made available, will be quite useful for benchmarking our S-$CO_2$ estimates, and reducing the amount of extrapolation required to make side-by-side comparisons.

## 2.3 Conceptual Layout of S-$CO_2$ Power Conversion Unit

Packaging an entire gas turbine power plant to fit into a single PCU vessel is, to say the least, a complicated process. To facilitate the process small scale (1 cm= 1m) physical models have been constructed. This ensures that all components, process streams, ducting and valve placement have been accommodated in a practical fashion.

Figure 2.3 is a photograph of the overall PCU model, but with a transparent pressure vessel to permit viewing of its internals. The recuperators are also removed so as not to obscure sight lines. The companion Figure 2.4 identifies the arrangement of all major components visible in the photograph.

Figure 2.5 is a photograph of a section through the S-$CO_2$ cycle's two recuperators, which fill the annulus outside the central cylindrical tube housing the turbomachinery. Also evident are the two partitions which separate the three low pressure plena: Turbine exhaust / high-T recuperator / low-T recuperator. Again we include a key to this layout in Fig. 2.6. One point worth noting is that the pressure vessel itself sees only low pressure gas (≈ 8 MPa): the high pressure gas (≈ 20MPa) is confined to the internals of the compressors, recuperators and the outlet plenum which feeds the duct returning the working fluid to the IHX / Reactor.

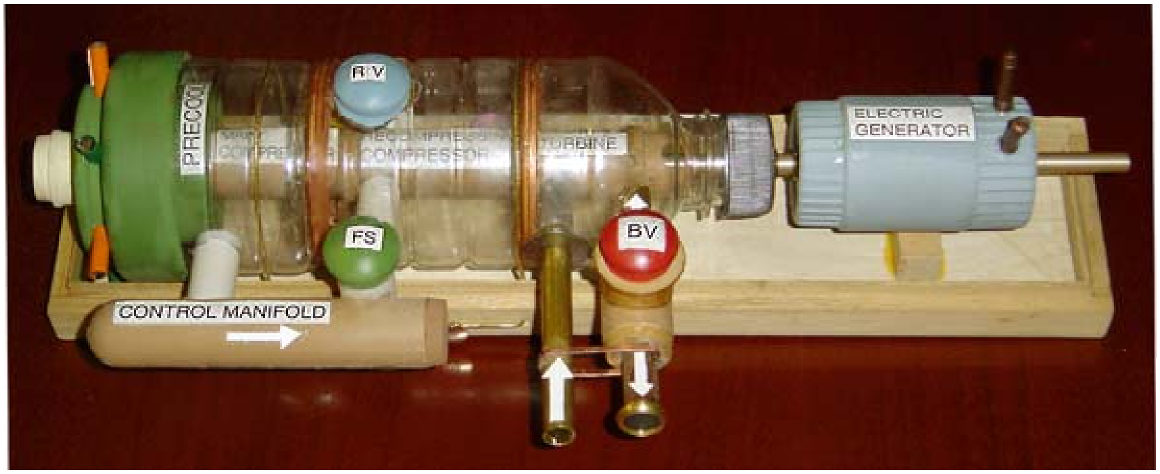


**Fig. 2.3 Photograph of PCU Model**
**(Transparent Vessel, Recuperator Cluster Removed)**

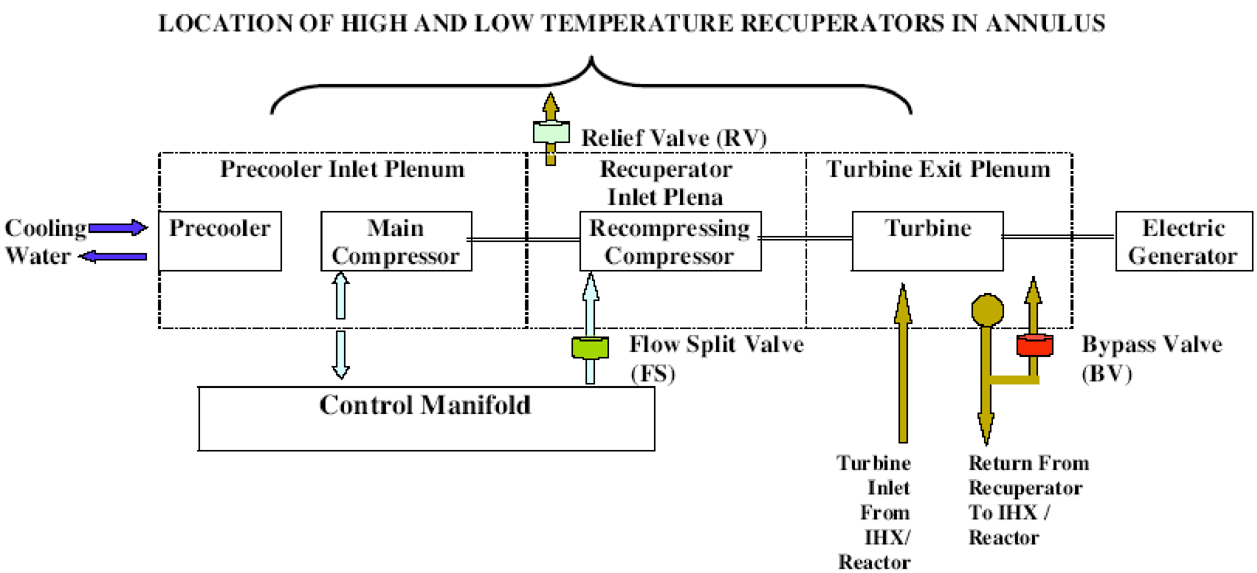


**Fig. 2.4 Key to Layout of S-$CO_2$ PCU Model**

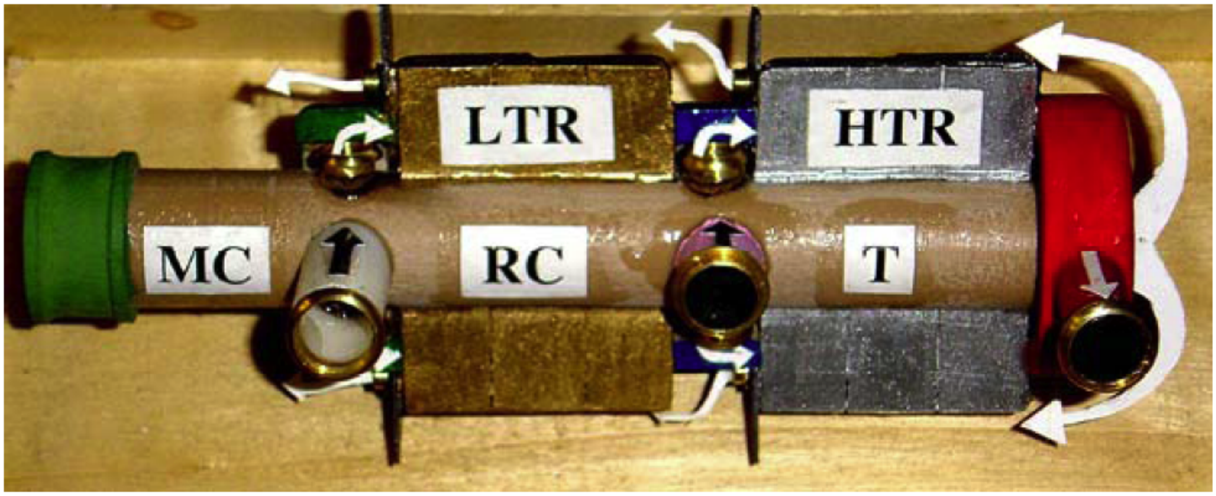


**Fig. 2.5 Photograph of Section View of S-$CO_2$ Recuperators (Surrounding Turbomachinery Nacelle)**

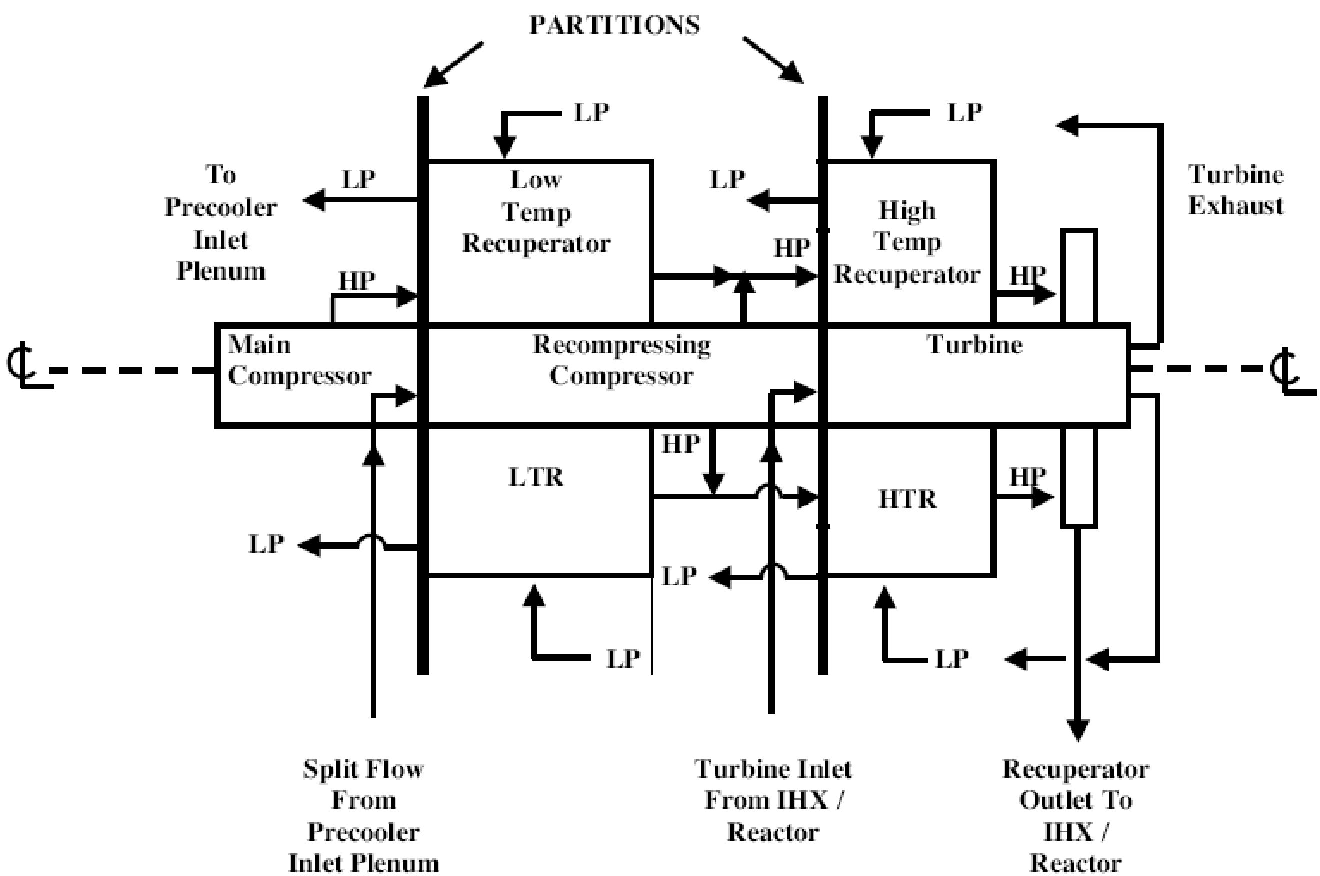


**Fig. 2.6: Key To Features in Recuperator Cutaway Section Model**

The basic features are very similar to those described by Dostal in his ScD Thesis / Topical report (Ref. 2.1): one should refer to this document for additional detail. His design in turn has much in common with the GA GT-MHR. Some particulars on the PCU vessel are as follows:

| Vessel Type | Dia. (m) | Height (m) | Operating Pressure (MPa) |
|---|---|---|---|
| **S-$CO_2$ PCU** | **7.6** | **12** | **8** |
| | **(including generator: 18 m)** | | |
| **PWR PV** | **5** | **13** | **15** |
| **BWR PV** | **7.4** | **21** | **7** |
| **GA GT-MHR PCU** | **8.7** | **37** | **NA** |

Thus the vessel in question appears to be within the range of those which are currently fabricable and transportable.

In the future we hope to apply computer codes to more elegantly model the details of the power cycle arrangement. It should also be noted that such details can be expected to evolve as design work progresses. For example, the bypass valve shown was located as per Dostal's recommendation; several other locations are conceptually possible, some of which are preferred by other (e.g. He) PCU designers. The location here is motivated by the fact that the return to IHX / Reactor and turbine exhaust streams are close in temperature (440 and 440ºC, respectively), which minimizes thermal shock during the rapid transient which actuation of the bypass valve initiates. Also note that improved (multiported, wavy channel) versions of Heatric™ PCHE are now available, which will reduce the required PCU vessel diameter.

## 2.4 Cost-of Power Projections

The approach followed for cost of power estimation was to start with definitive published analyses for similar systems and to proceed by changing only those cost category entries needed to transform the original design into (as close as practicable an approximation to) the S-$CO_2$ plant version. Coupled with estimates of the concurrent change in plant thermal efficiency, one can then estimate busbar costs in mills/kWhre.

Reference (2.2) reports results for two indirect cycle plants: one using helium as the primary coolant coupled to a Rankine secondary steam cycle; and the other employing a Brayton secondary power cycle with helium as the working fluid. Table 2.2 summarizes the busbar cost projections for these plants, taken from Ref (2.2) including those for other options which it documents, which provide additional perspective.

Dostal (Ref. 2.1) evaluates the difference between helium and S-$CO_2$ direct cycle power plants, and finds their busbar costs virtually identical: i.e. S-$CO_2$ / He = 1.004, which is well within the attendant imprecision of such estimates. From this we can infer the S-$CO_2$ indirect cycle value shown in row (9) at the bottom of the table, by also equating it to the corresponding entry in row (3) for its helium counterpart.

**Table 2.2: Summary of Projected Generation Costs ***

| | | Busbar Cost | |
|---|---|---|---|
| | **System** | **Mills/kWhre** | **Ratio** |
| 1. | Steam Cycle MHTGR (Ref. Case) | 51.0 | 1.00 |
| 2. | He Gas Turbine MHTGR, Direct Cycle | 40.4 | 0.79 |
| 3. | He GT MHTGR, Indirect Cycle | 49.0 | 0.96 |
| 4. | Advanced LWR | 44.2 | 0.87 |
| 5. | Coal, Pulverized | 48.9 | 0.96 |
| 6. | Coal, IGCC | 49.1 | 0.96 |
| 7. | Gas, CCGT | 48.9 | 0.96 |
| 8. | S-$CO_2$ ,Direct Cycle (Dostal) | 42.3 | 0.83 |
| 9. | S-$CO_2$ : Gas Turbine, He to S-$CO_2$ Indirect Cycle (inferred) | 49.0 | 0.96 |
| 10. | S-$CO_2$: Gas Turbine, liquid primary coolant (LPC) to S-$CO_2$ Indirect Cycle (estimated) | ≈ 46 | ≈ 0.90 |

* Entries 1-7 are from Ref (2.2)

The final entry in the table, row (10) displays an estimate for an indirect S-$CO_2$ cycle employing "LPC", a liquid (Na, LBE, FLIBE) as primary coolant, based on the observation made earlier that it would incur roughly half the efficiency penalty of a gas-to-gas indirect cycle. Reference (2.8), currently in press, provides relevant comparisons between ALMR Rankine and Brayton cycle units.

Based on these results several observations are of interest:

a. The S-$CO_2$ LPC version generates electricity about 10% cheaper than the reference case Rankine cycle MHTGR.
b. It is competitive with the fossil options cited.
c. It is more expensive than direct cycle GT concepts.
d. At this juncture it is not yet certifiably less expensive than an advanced LWR.

Point (a) is also supported by Dostal's plant efficiency comparisons for LBE cooled indirect cycle reactors reported in Ref (2.3) and reproduced here as table 2.3. The S-$CO_2$ version delivers about 5% more electricity than the Rankine option, with a further prospect for reducing mills/kWhre because the PCU costs less than a Rankine steam plant.

**Table 2.3 Net Efficiency Evaluation of LBE Cooled Indirect Cycle Reactors (Ref 2.3)**

| | Thermal power (MWth) | Cycle efficiency (%) | Gross electric power (MWe) | Self-consumption (%) | Net electric power (MWe) | Net efficiency (%) |
|---|---|---|---|---|---|---|
| Steam cycle | 700 | 42.7 | 298.9 | 7 | 278.0 | 39.7 |
| Helium cycle | 700 | 32.0 | 224.0 | 5 | 212.8 | 30.4 |
| Supercritical $CO_2$ cycle | 700 | 43.8 | 306.6 | 4.5 | 292.8 | 41.8 |

With respect to the last observation it should be noted that the LPC versions can accommodate breeder reactor cores, which utilize uranium a factor of fifty or more more efficiently than an LWR, and which are considered by many to be better minor actinide incinerators. Thus they have a better future prospect for sustainability – an important GEN-IV program goal. A second point is that the LPC results do not yet fully reflect all potential cost savings, in particular

a. A smaller containment structure can be employed because the primary coolant is not volatile, and the secondary plant can be located outside containment.
b. The LPC/S-$CO_2$ plants operate at maximum temperatures about 350°C cooler than helium cooled HTGRs, which reduces materials costs and the need for component cooling.
c. It is likely that the S-$CO_2$ turbine inlet temperature can be increased from the rather conservative value of 550°C.

A final observation is that essentially the same power conversion unit (PCU) can be used in both indirect and direct cycle applications. The latter version is competitive with the ALWR. This also suggests that the near term focus be on PCU cost quantification and reduction.

## 2.5 Component Cost Comparisons

Cost evaluations have been pursued at two levels: component-wise and on a complete power station basis (i.e. mills/kWre busbar costs). Both are based on relative comparisons versus better known alternatives. This section summarizes some of the more important component-wise findings; the preceding section deals with overall cost of electricity projections.

The most useful component cost scaling information found in the literature is that of Ref (2.4). As shown in Table 2.4 cost drops significantly with pressure for the principal components, which is advantageous for the S-$CO_2$ cycle compared to its helium counterpart. The lower temperature is less beneficial, as shown in Table 2.5, but a large advantage accrues from exploiting the increase in power rating per turboset.

**Table 2.4: Brayton Cycle Cost Scaling As a Function of Operating Pressure**

| | Cost Scaling Function | Ratio for 20 MPa / 8 MPa |
|---|---|---|
| **Recuperator** | $P^{-0.55}$ **(shell and tube)** | **0.60** |
| **Precooler** | $P^{-0.35}$ **(shell and tube)** | **0.73** |
| **Turboset *** | $P^{-0.6}$ | **0.58** |
| **Ducting** | **61 + P (MPa)** | **1.17** |

*** 1 shaft, 2 compression stages**

**Table 2.5: Brayton Cycle Cost Scaling**
**As A Function of Temperature and Power Rating**

| | Cost Scaling | Cost Ratio |
|---|---|---|
| **Recuperator** | **approx. 10% per 300˚C** | **~1** |
| **Precooler** | **~ constant with temperature** | **~1** |
| **Turboset *** | | |
| **Inlet T:** | $3.35 + \left(\frac{T_{\circ C}}{1000}\right)^{7.8}$ | **0.93 (650 vs. 850)** |
| **Specific power rating: cost per Mwe** | $W^{-0.32}$ | **0.56 (300 vs. 50 MWe)** |
| **Ducting** | $57 + \left(\frac{T_{\circ C}}{100}\right)^{2}$ | **0.77 (650 vs. 850)** |

*** 1 shaft, 2 compression stages**

It is important to note that the scaling relations developed by Schlenker were for a helium cycle. Nevertheless most trends should also hold for $CO_2$ working fluid. One major issue relevant to intercomparison between He and $CO_2$ is the much higher recuperator duty called for in the latter: about twice the MWth per MWe compared to the He cycle. However, in actual optimized cycles ($CO_2$ at 20 MPa, He at 7 MPa) the recuperator volumes, hence costs, are approximately equal.

Another factor to consider is that the higher temperatures of the helium cycle would require use of more expensive materials in an IHX. Reference (2.5) shows that heat exchangers constructed of Inconel 625 are more expensive than those made of 316L SS by the ratio 5.0 /2.2 = 2.3.

In terms of absolute costs for the Heatric™ units employed for the high temperature recuperator, low temperature recuperator, and (titanium) precooler, Dostal's results scale to about 7 million dollars each for a 250 MWe unit, plus another 4 millions dollars for an IHX. The total of 25 million dollars represents about ten percent of total plant capital cost for a target total overnight cost of 1000$/ kWe. However his results are for older straight channel Z-flow Heatric™ units. At a workshop at MIT on 10/02/03, Heatric™ representatives stated that their new multiported configuration could "cut cost by a factor of about two". Thus refinement of both heat exchanger design and cost estimates deserve high priority in future work at MIT and elsewhere.

Very little has been done to date to estimate turbomachinery costs. One promising approach is to scale costs from those of the high pressure turbine in supercritical steam cycles (Ref 2.6) which have turbine inlet conditions of 250 bars and 560°C—remarkably close to those of the S-$CO_2$ units at 200 bars and 550°C. Power ratings of 250 MWe are also comparable.

## 2.6 Indirect vs. Direct Cycles

Most earlier work at MIT on S-$CO_2$ Brayton power conversion has been directed toward direct cycle GFR applications. The present contract is exclusively focused on indirect cycle applications, which conveniently fall into two categories:

1. Reactors with sodium, lead alloys or molten salt as primary coolant
2. Reactors with helium as primary coolant.

Fortunately the power cycle itself is not affected to any significant extent by the nature of the heat source, whether reactor core or intermediate heat exchanger. Optimum thermodynamic state points and all component thermal hydraulics remain the same. This considerably facilitates comparisons among applications and, in particular, assessment of the effects of employing an indirect cycle. Since the two most advanced nuclear power plant designs using the Brayton cycle (with helium as the working fluid) are direct cycle—the Eskom PBMR and the GA GT-MHR—this observation is of considerable practical utility.

Use of an indirect cycle affects the unit cost (mills/kWre) of generating electricity in two main ways:

1. Reduced thermal efficiency because of the added primary coolant circulator power consumption and the reduction in turbine inlet temperature due to the $\Delta T$ needed to transfer heat across the IHX; and
2. Increased capital costs due to the added IHX and circulator (in some instances compensatory savings may accrue)

Efficiency Penalty of Indirect Cycles

Adding an intermediate loop between the core and power cycle reduces cycle efficiency through two effects: blower power consumption and reduced turbine inlet temperature. Approximate relations for these losses (derivable for ideal gas—ideally recuperated Brayton cycles) are:

$$\Delta\eta_w = (1-\eta_o)\frac{\Delta W_b}{Q} \equiv (1-\eta_o)\left(\frac{\Delta P}{\rho c_p \ \Delta T_c}\right)$$

$$\Delta\eta_T = (1-\eta_o)\frac{\Delta Th}{Th}$$

where $\eta_o$ = reference cycle thermodynamic efficiency
$\Delta W_b$ = primary circuit blower (circulator) power consumption, MWe
Q = core thermal power, $MW_{th}$
Th = turbine inlet temperature, ºK
$\Delta Th$ = reduction in Th due to added IHX heat transfer film drops.

For the two principal categories—liquid vs. gaseous primary coolant—a rough distinction can be made:

| | $\Delta T_h$ | $\Delta W_b/Q$ |
|---|---|---|
| Liquid | 20ºC | 0.005 |
| Gas | 40ºC | 0.02 |

Thus for a S-$CO_2$ system having $T_h$ = 820ºK and $\eta_o$ = 0.44, $\Delta\eta_w$ =0.0112, while $\Delta\eta_T$ = 0.0273 for a combined efficiency loss, $\Delta\eta$ =0.0385 or about 4%, which more detailed simulations confirm. A comparable liquid cooled primary system will, for the above parameters, have a $\Delta\eta$ of about 1.65%. This latter value compares well with the 1.43% loss predicted in Ref (2.3), for lead-bismuth-eutectic primary coolant, considering the crude nature of the analysis.

If capital costs (in \$) were unaffected, the increase in busbar unit cost e, mills/kWhre would be

$$\frac{\Delta e}{e} = -\frac{\Delta\eta}{\eta_o}$$

Hence a $\Delta\eta$ of –2.2% would increase mills/kWhre by 5%.

It should be recognized that the circulator power consumption W and the heat transfer film temperature differential $\Delta T_f$ are not independent variables. One finds that for a given service and heat exchanger design, the product of W and $\Delta T_f$ is approximately constant. Hence a tradeoff can be performed to minimize total $\Delta\eta$. If one broadens the scope to include heat exchanger design, one finds that the product of W • $\Delta T_f$ times frontal flow area squared is approximately constant (2.7). An iterative procedure of this sort is well worthwhile, but will not be pursued further here. In any such analyses one must use more sophisticated cycle optimization codes, especially when dealing with non-ideal gases like $CO_2$ near its critical point. For example, Dostal has reported that S-$CO_2$ efficiency is fairly insensitive to reactor core or IHX pressure drop: for example an increase from an already generous 250 kPa to 500 kPa reduces $\eta$ from 44.75 to 44.25%. He also finds that a 40ºC decrease in turbine inlet temperature reduces cycle efficiency by about 1.8%, considerably less than the 2.73% predicted by our simple ideal gas/ideal cycle model.

## The Path Forward

Most of our focus to date has been on capital cost reduction, since this category typically accounts for about 70% of the cost of nuclear-generated electricity. However, reducing capital cost by a factor $(1 + \delta)$ has the same effect on capital-related mills/kWre as increasing capacity factor by the same ratio. Thus attention must also be paid to factors such as ease of inspection and on-line maintenance when developing a plant layout. This has motivated development of the modified power conversion unit described in the chapter which follows.

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# Chapter 3 Alternative Power Conversion Cycle Arrangement

## 3.1 Introduction

An alternative “unbundled”, plant layout has been devised.

In the preceding chapter we showed an integral PCU vessel containing all components: similar to that employed in the GT MHR and by Dostal in his conceptualization of the S-$CO_2$ power cycle.

Concern over access for inspection and maintenance has inspired further work to unbundle the power plant into two vessels: one containing all heat exchangers and the other being the turbomachinery nacelle. Figure 3.1 is a picture of the model assembled to work out arrangement details and Fig. 3.2. is a key to component layout. Another advantage of the new arrangement is that a modest reduction in the largest vessel diameter can be realized. We currently estimate 5 m, which is the same as that for a large PWR--hence guaranteeing fabricability and transportability. The principal detrimental aspect is that the vessel now has six gas-duct wall penetrations as opposed to four with the prior monolithic all-in-one design. In the monolithic design as few as one penetration for ducts might be realized by using a coaxial inlet/outlet duct and by encapsulating all valves inside the PCU vessel.

Another change of note is relocation of compressor flow split control valves to the compressor outlet, where fluid density is much higher, and also where added pressure loss will not move the $CO_2$ state point closer, or even into, the two phase region. This change applies to both types of layout.

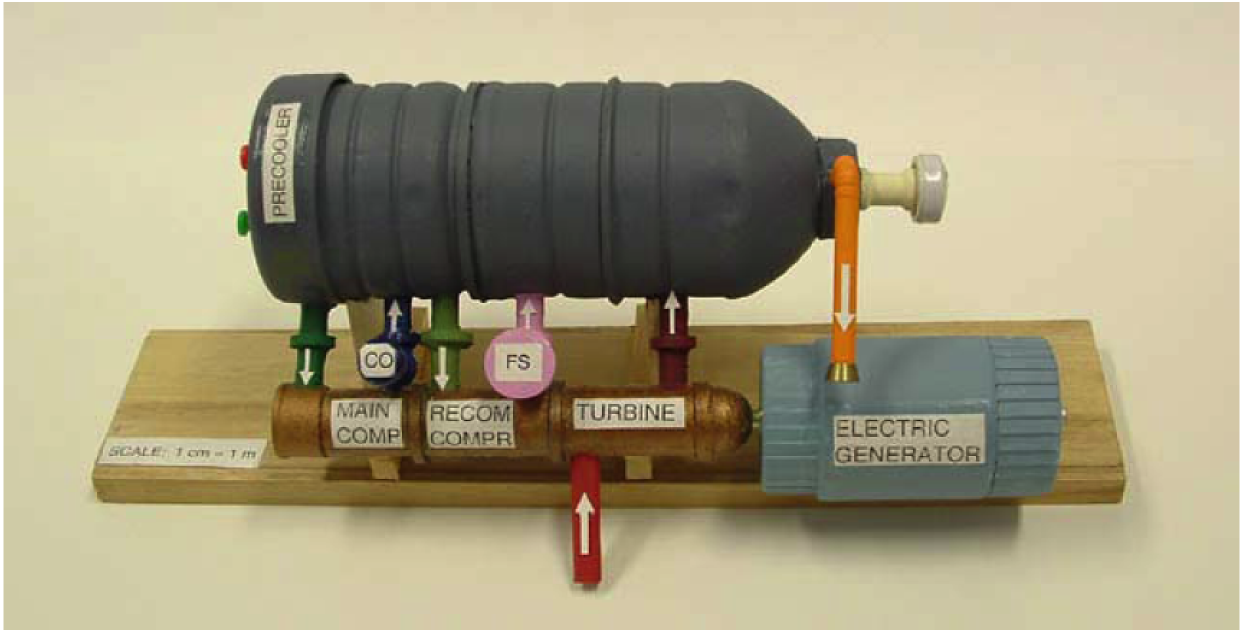


**Fig. 3.1 Model of Two-Vessel Plant Layout**

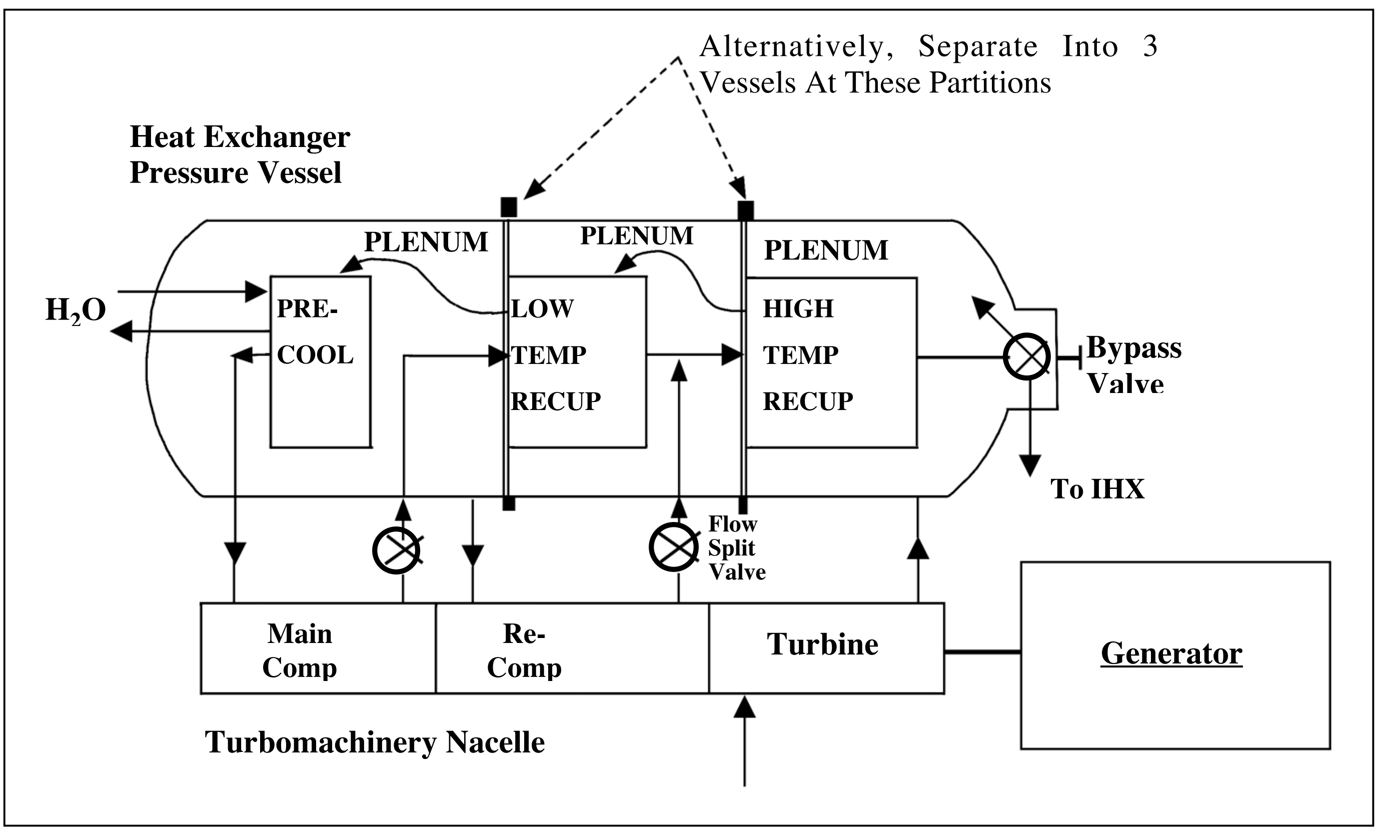


**Fig. 3.2 Unbundled Two to Four Vessel Power Cycle Layout**

## 3. 2 Discussion of Design Issues

One challenge to any multi-vessel layout will be accommodation of thermal expansion, and the resulting stresses, in the crossducts connecting the vessels and components. At the expense of added pressure drop, one can employ “S” or “Ω” shaped crossducts. A more exotic approach would be the use of INVAR type alloys (3.1), which have thermal expansion coefficients that are a factor of two or three lower than those of conventional steels below about 400°C.

The first resort, however, should be use of internal insulation inside both ducts and vessels. This will lead to cooler operating temperatures of pressure-bearing surfaces, and smaller heat losses—both of which are beneficial. All gas-cooled reactors share this problem; hence we should be able to exploit a useful reservoir of past experience and the fruits of current R&D (3.2). For the present, the straightforward approach of using nested thin annular metal shells separated by thin gas gaps appears attractive, since this is a proven means for reducing both radiation and conduction. Similar results can be achieved using less rugged materials/configurations such as ceramic wool or foam (but not solid ceramics, which have thermal conductivities roughly two orders of magnitude higher than gases).

We are favored in this regard since $CO_2$ has a factor of six or so lower thermal conductivity than He, and our S-$CO_2$ cycle has a factor of roughly two lower $\Delta T$ between turbine inlet temperature and ambient. Nevertheless, crossduct stress analysis, both steady state and transient will play an important role—especially for the 440°C turbine outlet line; the other crossducts pose less of a problem: main compressor inlet at 32°C, outlet at 61°C; recompressing compressor inlet at 66°C, outlet at 153°C. The longer hot ducts from and to the IHX (at 550°C and 440°C) are easier to accommodate. At present we do not see any benefit to making them coaxial, as is common for direct cycle helium Brayton designs.

## 3.3 The Path Forward

The least defined aspect of what we will henceforth designate as the “Mark II” layout is the arrangement of heat exchangers inside their pressure vessel. Most of our Heatric™ PCHE analyses to date have been for their older Z-flow configuration. It is important to upgrade this work to employ their recent multiported (MP) design which is completely countercurrent, and can employ zigzag (sawtooth) channels, leading to factor-of-two reductions in volume and cost.

In a related development we have completed arrangements to purchase a 24 kW recuperator from Heatric™ for late summer delivery. Tests on this unit will help validate our computer models for HX design. It is our understanding that ANL is also planning performance tests of a PCHE for $CO_2$ /$CO_2$ and $CO_2$ /$H_2O$ heat exchange (3.3).

Should vessel size and transportability prove to be an unexpected problem, we could also consider using a prestressed cast iron vessel (PCIV) (3.4). These vessels are assembled on site from modular segments, and their German developers claim many beneficial aspects, not the least being reduced cost compared to conventional welded forged steel pressure vessels.

Other arrangement uncertainties can only be resolved once plant control and transient response studies are well underway, since several alternative locations for bypass valves are possible (3.5) as are the number and location of valves required to control the flow split between the main and recompressing compressors.

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